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DRAWINGS ATTACHED.

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COMPLETE SPECIFICATION.

Improvements relating to Blading of Axial Flow Turbines.

We, MASCHINENFABRIK AUGSBURG-NÜRNBERG A.G., a German Company, of 7 Stadtbachstrasse, (13b) Augsburg 2, Germany, do hereby declare the invention, for which we pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:—

The present invention relates to blading for axial flow turbines and in particular to blading for gas turbines and is concerned with the provision of improved blading for such turbines.

In the design of blading for gas turbines, a large number of design rules have been developed. The choice of any particular rule is not necessarily influenced by the experimental data available to the particular designer.

The design rule most frequently used in turbine construction is the rule of the free vortex flow, namely $c_u \cdot r$ = constant where c_u is the peripheral component of the absolute velocity of the working fluid, which may be calculated from the adiabatic pressure drop and r is the distance from the axis of blade rotation of the blade portion under consideration. This results in laminar flow and gives very favourable fluid-flow conditions over a wide range of blade height and for considerable differences in the inner and outer pitch of the blading. As will be explained later, in the channels of twisted blades, however, under the conditions which generally obtain in practice, the fluid-flow conditions in the vicinity of the blade drum are worse than those in the channels of cylindrical blades. The term "cylindrical blade" is used herein to define a blade having parallel generating lines and having a blade cross section of aerofoil form. If therefore, the hydraulic efficiency at various radial points on a blade are tabulated, it will be seen that the cylindrical blade has the best, as it were, "local" efficiency in the vicinity of the blade drum, whereas the twisted blade gives better results in the radially outer regions of the blade.

According to the present invention there is provided blading for an axial flow turbine, more particularly a gas turbine, wherein at least each rotor blade has a radially outer portion, which is so twisted that in operation of the turbine the working fluid leaving this blade portion obeys at least approximately the law of free vortex flow, namely $c_u \cdot r$ = constant, and a radially inner portion which is formed by parallel generating lines, so that the working fluid leaving this portion obeys at least approximately the relationship $c_u \cdot r^{\cos^2 \alpha_1}$ = constant, where, in both causes c_u represents the peripheral component of the absolute velocity of the working fluid, r the distance from the axis of blade rotation of the blade portion under consideration and α_1 the angle between the direction of blade velocity and the direction inlet flow of the working fluid.

If the twisted blade portion is designed for the same pressure gradient at the same speed as the cylindrical blade portion, that is to say, for $\frac{u}{c_0}$ = constant, where u is the blade peripheral velocity and c_0 the velocity of the working fluid calculated from the adiabatic pressure drop, the functions (i.e. the direction and magnitude of the velocity of the working fluid, degree of reaction and static pressure) which characterise the local flow conditions can be so selected in the transition zone between the two dif-

ferent blade portions that the two flows do not interfere with each other, and therefore no layer of separation is formed. If the twisted and cylindrical blade portions are designed for the same degree of reaction at the transition zone, an equal static pressure is obtained either side of the zone. For ease of manufacture, the transition from one blade portion to the other is made smooth, that is to say, without a sharp bend.

Compared with the known designs, a blade of the invention has the following advantages:—

For the static and dynamic strength of a turbine rotor blade more particularly a gas turbine rotor blade, the modification in profile cross-section along the blade length or the configuration of the blade is very important. As the stage pressure gradient is generally given beforehand, the smallest $\frac{u}{c_o}$ is desirable for strength reasons. For a constant direction of gas flow from the blade channel the degree of reaction is substantially dependent on $\frac{u}{c_o}$. Furthermore, as the peripheral speed "u" near the blade drum is lower than at the blade tip, a blade which is twisted over its whole length, must have, in the vicinity of the blade drum, a profile with a large deflection or deviation in addition to a large cross-sectional area. The fluid flow conditions in the vicinity of the shape of the blade channel does not produce an acceleration of the working medium, which is necessary to avoid losses. If however, in the drum vicinity, the blades are made cylindrical, conditions become more favourable. To maintain a shockless admission to the blade channel, the stator blades are expediently made cylindrical, at least over the same radial extent. They could, however, also be made twisted over this range, for instance for achieving a desirable distribution of the meridian velocities.

It must also be borne in mind that in the case of twisted blade design, the rule of free vortex flow applies, that is, the whirl in the gap between a stator and the rotor varies according to the formula

$$c_u \, . \, r = \text{constant},$$

while in the case of cylindrical blading, on the contrary, the whirl varies according to

$$c_u \, . \, r^{\cos^2 \alpha_1} = \text{constant},$$

where α_1 is the angle between the direction of blade velocity and the direction of inlet flow of the working fluid. In the latter case, therefore, the whirl in the gap varies less over the radius. This means that for a given reaction in the middle portion, the degree of reaction also varies less over the radius, which is generally desirable. If, on the other hand, the degree of reaction at the blade drum is known beforehand, possibly by the critical pressure gradient in the stator, the blading of the invention gives a lower degree of reaction, that is to say, lower peripheral velocities for the same stage pressure gradient, i.e. lower blade stressing. As is known, the losses due to leakage through the gap at the blade tip are proportional to the degree of reaction. Since this degree of reaction in a blading of the invention is smaller than in a case of blades twisted over the entire length, also these losses are reduced.

In order that the invention may be more clearly understood, one constructional example will be explained in the following with reference to the accompanying drawings, in which:—

Figure 1 shows a rotor blade of a turbine in side view;

Figure 2 shows a series of profile sections through the blade of Figure 1 in the planes A—A; A^1—A^1; B—B and C—C respectively;

Figure 3 is a longitudinal section through the blade of Figure 1 on the line III—III; and

Figure 4 shows the velocity triangles of the blade shown in Figures 1 and 2.

Referring to Figures 1 to 3, 1 denotes a rotor blade and 2 a blade drum carrying the blade. The portion of the blade adjacent the drum is formed cylindrically from A to A^1, i.e. the generating lines of this blade part run parallel to each other. The outer portion, on the contrary, from A^1 to C, is twisted. The transition zone from one blade portion to the other, indicated in Figures 1 and 3 by the broken parallel lines as "z" is rounded off, so as to give a continuous transition, which is favourable from the point of view of manufacturing and flow techniques.

Referring to Figure 4 in which "u" denotes the vector of the blade peripheral velocity, and O—A represents the blade velocity at the section A—A, O—A^1 represents the blade velocity at the section A^1—A^1, O—B represents the blade velocity of the section B—B and O—C represents the blade velocity at the section C—C, i.e. the blade peripheral velocity. The vectors of the velocities of working fluid relative to the blade are denoted by "w" and those of the absolute velocities of the working fluid by "c". The suffix 1 denotes the inlet side and the suffix 2 the outlet side. The suffixes A, A^1, B and C likewise relate to the cross-sections A—A, A^1—A^1, B—B and C—C. α_1 is the angle between the direction of

blade velocity and the direction of inlet flow of the working fluid.

WHAT WE CLAIM IS:—

1. Blading for an axial flow turbine, more particularly a gas turbine, wherein at least each rotor blade has a radially outer portion, which is so twisted that in operation of the turbine the working fluid leaving this blade portion obeys at least approximately the law of free vortex flow, namely $c_u \cdot r$ = constant, and a radially inner portion which is formed by parallel generating lines, so that the working fluid leaving this portion obeys at least approximately the relationship $c_u \cdot r^{\cot^2\alpha_1}$ = constant, where, in both cases, c_u represents the peripheral component of the absolute velocity of the working fluid, r the distance from the axis of blade rotation of the blade portion under consideration and α_1 the angle between the direction of blade velocity and the direction of inlet flow of the working fluid.

2. Blading as claimed in Claim 1, including a continuous transition zone between the inner and outer portions of each blade.

3. An axial flow turbine having rotor blading as claimed in Claim 1 or Claim 2 and stator blades of a form similar to that of said inner portions over the same radial distance as said inner portions.

4. An axial flow turbine having rotor blading as claimed in Claim 1 or Claim 2, and stator blades twisted over their entire blade length.

5. On or for an axial flow turbine blading substantially as hereinbefore described with reference to the accompanying drawings.

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908,748 COMPLETE SPECIFICATION
1 SHEET

This drawing is a reproduction of the Original on a reduced scale.

Fig.1

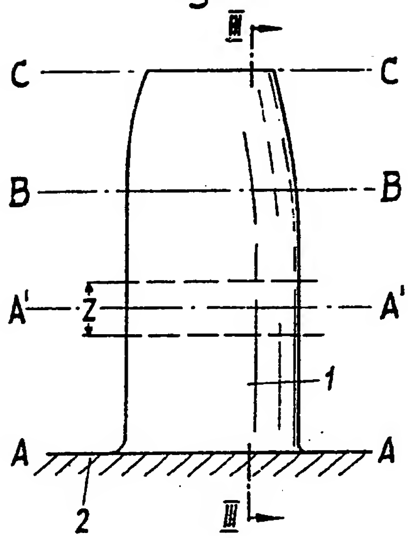

Fig.2

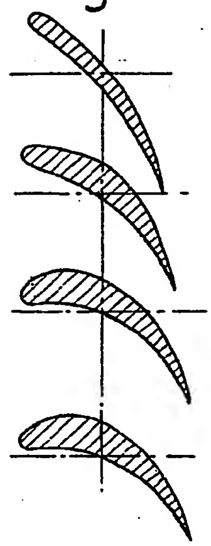

Fig.3

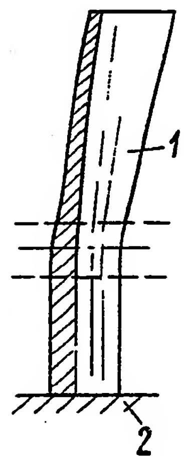

Fig.4